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POSSIBLE MATERIALS NEEDS FOR ENERGY ABSORPTION IN

SPACE-VEHICLE LANDINGS

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POSSIBLE MATERIALS NEEDS FOR ENERGY ABSORPTION IN
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SUMMARY

In the design of systems to absorb the landing impact energy of space vehicles, such as the system which will be used to protect the Lunar Excursion Module of the Apollo spacecraft in touching down on the moon, the specific energy absorption (energy absorbed divided by the weight of the material deformed) is not necessarily the primary factor. Other factors which must be considered include the compatibility of the deforming structure with its support, the packaging volume of the system, the control of penetration into the landing surface, and the desirability of the system with respect to tip-over stability and maximum permissible acceleration. These factors can lead to consideration of systems which do not employ energy absorbers of maximum efficiency. Materials improvements leading to better energy absorption in these systems could result in significant gains, particularly for the systems having relatively inefficient absorbers. In addition, it is recommended that materials be improved by decreasing their elastic bounce-back and increasing their resistance to the effects of the rocket exhaust and the space environment. Such materials improvements may play a major role in the selection of energy absorbing systems.

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INTRODUCTION

It is generally accepted that an energy-absorbing system will be employed for so-called "soft" landings on the moon (refs. 1 and 2), and such a system would manifestly be required for parachute landings on planets. For the lunar landing, it would be desirable to design the system to absorb higher energies than commonly expected because of unforeseen problems that may occur when the vehicle is hovering near the landing surface. For example, the possible disfiguration of the lunar surface by the rocket exhaust, together with any resulting damage to the vehicle by dislodged lunar particles, could conceivably require premature shutdown of the rocket engine. In addition, there is always the possibility of engine failure while hovering.

Important differences exist between energy absorption for space vehicles and conventional aircraft. The large temperature gradients and hard vacuum of the space environment suggest that, in order to be reliable, the energy absorbing system should be simple. Because many space-landing systems need never be used more than once, a controlled failure of the mechanism can be employed, and a failure mechanism is inherently simpler than a mechanism permitting repeated use.

Considerable effort has been devoted to systems for absorbing the landing energy of spacecraft, as indicated by the representative sampling in references 3 - 18. The systems described in references 3 - 18 generally have three or more legs, but a few employ single gas bags.

Another area of effort pertains to the energy-absorbing mechanisms within the systems, and references 19 - 30 bear on this area (where refs. 23, 24, and 28 are regarded as dealing with mechanisms rather than systems because they do not concern themselves with tip-over problems arising from unsymmetrical

landing conditions). In reference 27, Esgar surveys the many energy absorbing mechanisms available and classifies them as follows:

- (1) Materials deformation (honeycomb, balsa, plastic foam, frangible tube, extrusion, wire drawing)
- (2) Gas compression (air bags)
- (3) Rockets (which may fail or have to be turned off, as indicated above)
- (4) Friction and mass acceleration (spike, frangible tube, broaching, etc.)

In the present paper, attention will be restricted to materials deformation mechanisms. The first two major sections of the paper both lead to observations on the importance of mechanism energy absorption in the selection of mechanisms for over-all systems. In the first section, the mechanism selection is based on over-all energy absorption alone while, in the second section, it is based on a balance between energy absorption and several important supplementary requirements. In the third major division, the observations from the first two sections are applied to yield a conclusion on the range of mechanisms for which materials improvements are needed; and some general comments are made, with specific examples, on the nature and possible effect of these needed improvements.

NOTATION

c	spacing parameter of reversible tube, dimensionless
D	inside diameter of reversible tube, in.
g	acceleration of gravity on earth, 32.2 ft per sec ²
K	ultimate strength in shear, lb per in. ²
P	crushing load on reversing tube, honeycomb core, and over-all strut
P _B	buckling load of over-all strut
s	cell size of honeycomb from flat to flat, in.

SEA specific energy absorption, defined as energy absorbed in foot-pounds divided by the weight of material deformed in pounds

t thickness of honeycomb material, or reversible tube wall, in.

U energy absorbed by reversing tube

V vertical impact velocity, ft per sec

x stroke of energy absorber, in.

α ground slope, deg

ρ density of a material, lb per in.³

σ_y effective yield stress, lb per in.²

ENERGY ABSORPTION

In order to show how mechanism energy absorption affects the selection of mechanisms for optimum over-all energy absorption, the first step herein will be to describe two mechanisms and rate them according to their SEA (see Notation). The second step will be to incorporate the mechanisms into initial design estimates and rate the results according to calculated values of over-all efficiency, which is energy absorbed divided by the weight of the complete mechanism plus its supporting structure. The third step will be to compare the ratings according to SEA and over-all efficiency and evaluate the significance of the comparison.

Energy-Absorbing Mechanisms

As a start on the first of the three steps listed above, a description is given of an inverting tube mechanism (ref. 26). This mechanism employs soft aluminum alloy or mild steel and absorbs energy as it inverts or "turns inside out." An analytical prediction of the inverting load has been

undertaken at the Ames Research Center, with results shown in Fig. 1. The die, shown dotted, transmits the constant load, P , and inverts the tube through a distance x . The tube has an inside diameter, D , and a wall thickness, t , and the spacing during inversion is ct . As indicated in Fig. 1, energy is absorbed by plastic bending, compression, and shearing, plus friction. Elastic effects are negligible. For the actual analysis, shearing and friction are also neglected, giving a total energy, U , composed of bending and compression. After U is evaluated, it leads to the expression at the bottom of Fig. 1 for the load, P , divided by the effective yield stress, σ_y .

In Fig. 2, the results of Fig. 1 are expressed in terms of SEA/σ_y . Also shown in Fig. 2 are the calculation of the spacing, c , for minimum energy and, at the bottom of the figure, the corresponding value of SEA/σ_y . For the determination of σ_y , a greatly simplified von Mises yield criterion is used (ref. 31), and this criterion is given in Fig. 2, where K is the shear ultimate.

Results are shown in Fig. 3 for a 3003-H14 aluminum tube, which has a K value of 14,000 lb per sq in. (ref. 32). The upper solid line is a plot versus t/D of the reversing tube SEA in ft-lb per lb, as minimized with respect to spacing. The dashed line is a fairing of experimental points taken from reference 26 and adapted to the present terminology. Agreement with the reversing tube theory is seen to be good.

The lower solid line and the t/s notation in Fig. 3 apply to hexagonal honeycomb, which is the second energy absorbing mechanism to be described. Here s is the cell size from flat to flat, and t is the thickness of the material (giving wall thicknesses of t on four sides of

a cell and 2t on two sides). For a crushing stroke parallel to the cell axes, a theory has been devised in reference 29 and found to compare favorably with experiment. In Fig. 3, the results of reference 29 have been converted to SEA values for 3003-H18 aluminum alloy. This conversion includes the effect of stroke limitation due to compacting, as estimated from information in references 25 and 29.

It should be noted that the maximum SEA values in Fig. 3 correspond to practical upper limits in t/D or t/s , above which the particular mechanisms and materials fail to function. On this basis, it can be seen that the present honeycomb core has a lower maximum SEA than the present reversing tube. Such will not be the case in general since the reversing tube is restricted to a softer metal than is the honeycomb core. The SEA rating in Fig. 3, however, will constitute a useful comparison after the two mechanisms and materials which result in the ratings have been incorporated into initial design estimates.

Application of Energy Absorbing Mechanisms

In the proposed design estimates, the two mechanisms of Fig. 3 are utilized in alternative energy absorbing struts, of identical length, for a hypothetical lunar landing vehicle. The general landing conditions of this vehicle are shown in Fig. 4, namely, an earth weight of 20,000 lb and a maximum acceleration of 6 earth g essentially in the direction of the four main energy absorbing struts. If all four pads impact simultaneously, these numbers result in a 30,000 lb maximum crushing load in each strut.

The geometry of a single strut is shown in Fig. 5 with its supporting structure and foot pad. The strut length is 9 ft, prior to crushing, the clearance is 6 ft, and a 4-ft stroke is selected. This stroke should, for a maximum load of 30,000 lb per strut, handle symmetric vertical landings

with impact velocities up to 30 or 35 ft per sec, depending on the extent to which the force-displacement curve deviates from a rectangular shape.

The two energy absorbing mechanisms described earlier, then, are incorporated into initial design estimates for a single strut having an undeformed length of 9 ft and carrying a constant 30,000 lb crushing load over a 4-ft stroke. The load is constant, giving a rectangular force-displacement curve, because the mechanisms are not tapered. Another ground rule requires the buckling load to be at least 90,000 lb, or in other words, at least three times the crushing load, and this factor is assumed to account for dynamic onset problems. Finally, the reversing tube mechanism is designed to provide its own resistance to column buckling while the honeycomb core is supported by tubing made of 6061-T6 aluminum.

It is considered that the ground rules just described are satisfactory for present comparison purposes, and specifications of this nature must be made to permit the determination of minimum strut weights. These minimum weights are determined by means of the curves in Fig. 6, which are plots of various component weights in pounds versus t/D for the reversing tube or t/s for the honeycomb core.

The reversing tube weight is defined by the upper solid line in Fig. 6. The intersecting vertical dashed line defines the buckling boundary for the reversing tube. To the left of the vertical line, the buckling criterion is satisfied since $P_B/P > 3$, where P is the crushing load for the present strut (30,000 lb) and P_B is the buckling load. To the right of the vertical line, the buckling criterion is not satisfied since $P_B/P < 3$. Since the upper solid line shows that the reversing tube weight, which is also the total strut weight, decreases as t/D increases, the minimum weight for the strut lies at the intersection of the tube-weight curve and the buckling boundary.

The total weight for the honeycomb strut is defined by the lower solid line in Fig. 6. The ordinate values for this curve are the sums of the ordinates for the nearly horizontal dashed lines, which represent the individual weights of the honeycomb core and its supporting tube. The entire curve for the supporting tube is defined by the buckling boundary, $P_B/P = 3$. Hence, the minimum total weight for the honeycomb strut is simply the minimum ordinate for the lower solid line, as marked by an intersecting vertical dash.

It can be seen in Fig. 6 that the minimum total strut weight is greater for the reversing tube than for the honeycomb core. In this connection, it should be recalled that the two systems are identical except for their struts and are required to absorb the same energy. Thus the strut-weight comparison just given means that the honeycomb system has a higher over-all efficiency than the reversing tube system.

Comparison of Ratings

It is useful, at this point, to compare the SEA ratings given earlier with the over-all efficiency ratings just derived from Fig. 6. The somewhat surprising result is that the mechanism producing the higher over-all efficiency, namely, the honeycomb core, has the lower value for its maximum SEA.

As a start toward an explanation of this reversal of SEA ratings, it is noted that two factors other than SEA have a large effect on over-all efficiency in the present example. The first factor is the addition of local structure necessary to contain or support a mechanism. Its effect is shown in Fig. 6 at the optimum t/s for the honeycomb strut, where the supporting tube weighs well over twice as much as the honeycomb core. The

second factor is the application of a mechanism below its maximum SEA value. In the present case, the optimum SEA values employed are as much as 42 percent lower than the corresponding maximums in Fig. 3. The cause of such a reduction can be seen in Fig. 6 in that the optimum values of t/D and t/s , which determine the optimum values of SEA, do not correspond to minimum weights for the reversing tube and honeycomb core mechanisms. It has already been noted that the optimum t/D and t/s values are determined by the buckling boundary of $P_B/P = 3$, and this means that the second factor is associated with buckling in the present example.

With two factors established which could conceivably reverse the SEA ratings in the present design comparison, a question remains as to why they do so. The fundamental answer is that the specified over-all design geometry calls for a rather long energy absorbing strut; and this length is an overwhelming disadvantage with respect to buckling for the mechanism having the higher maximum SEA, namely, the reversing tube. This buckling disadvantage occurs because the reversing tube supplies its own supporting structure and must be made of soft material, whereas the honeycomb core can be enclosed by hard tubing. With respect, then, to the two factors listed earlier, the first factor is misleading in that the supplying of its own support by the reversing tube should apparently save weight, but actually costs weight because of the buckling effect associated with the second factor.

At this point, a question naturally arises as to whether a reversal of SEA ratings is likely to occur in design comparisons other than the present example. The likelihood seems fairly great as long as the two factors which can reverse the ratings have broad applicability and as long as the over-all design geometry can represent a major advantage for one of the mechanisms being compared. The broad applicability of the first factor, namely, addition

of supporting structure, is self-evident, with examples being dies for frangible tubes, bases for crushing plastic or metallic foam, and clamps for ductile wires, together with tubes for enclosing crushable cores as considered herein. As for the second factor, it is hard to imagine a design which does not call for a compromise between the efficiency of the mechanism and the efficiency of its supporting structure. Finally, the choice of over-all design geometry will often be virtually specified by a variety of requirements, ranging from the absorption of prescribed energy to the use of existing hardware; and many of the energy absorbing mechanisms available are sufficiently intricate to be sharply advantaged or disadvantaged by specified geometry.

The foregoing discussion leads to the expectation that the present reversal of SEA ratings is not an isolated example, and such a conclusion manifestly diminishes the importance of the SEA in the selection of mechanisms for optimum efficiency of energy absorption in over-all systems.

INCORPORATION OF SUPPLEMENTARY REQUIREMENTS

Up to this point, a high over-all efficiency of energy absorption has been the only requirement considered in evaluating the importance of the SEA. Several supplementary requirements must also be considered, however, and, in fact, the requirement of satisfactory tip-over stability is important enough to justify a major decrease in over-all efficiency. A relatively small additional decrease may be justified by supplementary requirements other than stability. The last statement implies the premise, of course, that a system can be developed which is stable in tip-over but incorporates several other requirements while retaining a reasonably high over-all efficiency with respect to stable systems developed previously. The substantiation of this

premise and the consideration of its effect on the importance of the SEA comprise the major objectives in the present section of the paper.

The proposed substantiation is restricted to the lunar landing, and the first step is to describe the supplementary requirements considered herein and the model landing system developed to meet those requirements. The second step is to present experimental results for the model system and to discuss the extent to which those results actually substantiate the premise in question.

Supplementary Requirements and Model Landing System

With respect, then, to the first step given above, the supplementary requirements to be considered are the following: The landing system must

- (1) Provide minimum original packaging volume,
- (2) Reduce the possibility of excessive penetration of a potentially low-strength lunar crust,
- (3) Provide a stable firing platform for return blast-off,
- (4) Prevent the vehicle from tipping over as a result of unsymmetrical landing conditions, and
- (5) Provide an efficient shape for the boundary between acceptable and unacceptable impacts, as defined by tip-over and the maximum permissible acceleration.

Requirements (1) and (2) have led to the choice of plastic foam to absorb most of the prescribed impact energy. With respect to requirement (1), experience on earth suggests that foam should be foamable in flight after earth exit. This would result in a very low volume of material in the energy absorbing system during earth exit. An efficient distribution of this small volume would then require a minimum of fairing material for smooth, buffet-free air flow, with a resultant saving in weight.

Requirement (2) derives from the controversy over the depth of fairy castles or other low-strength lunar crust (as in refs. 33 - 47, where refs. 39, 42, 43, and 47 contain the principal advocacy of fairy castles). Since future unmanned probes may leave the controversy unresolved, it would seem necessary to accommodate requirement (2) as far as possible within weight limitations. Foam does this excellently because it permits a long stroke and a large bearing area without causing a major weight penalty or excessive packaging problem. Both a long stroke and a large bearing area tend to reduce the chance of penetrating the lunar crust to such a depth as to hamper exploration of the surface or return blast-off.

The third requirement has led to the choice of a three-legged landing system, with the result that no mechanical cranking of the legs is required to provide blast-off stability after landing. The prevention of tip-over specified in the fourth requirement refers to the moment of impact and calls for a landing system having wide outreach and low bounce-back. Finally, the boundary between acceptable and unacceptable impacts in the fifth requirement tends to be inefficient, that is, too restrictive at certain impact velocities and uselessly unrestrictive at others; and the force-displacement curve of the system should be adjusted to avoid such inefficiency.

The model energy absorbing system developed to meet the above requirements is shown in Fig. 7. This is the final version evolved from a long series of tests and studies and is shown undeformed. Three "ski-pole-type" cutters rest on three hollowed-out pieces of polystyrene foam, having a density of 1.8 lb per cu ft, and the cutting of the foam absorbs the largest part of the impact energy. Cutting is employed instead of crushing in order to reduce bounce-back and thereby improve tip-over stability. The clips shown on the cutters in Fig. 7 act like "fish-hook-type" barbs, once the foam has

been cut, and thus reduce bounce-back still further. Also shown in Fig. 7 are the horizontal tension members, the curved compression members (which buckle to absorb energy after the foam has been cut), the crimps at the top of the compression members (to simulate pinned joints for folding the system), the horizontally and vertically oriented accelerometers, and a 5-in. scale.

A picture of a complete model after impact is shown in Fig. 8, together with a portion of the free-fall testing facility. The landing surface on which the model rests is a 1-1/2-in. thick layer of crushed basalt. It can be seen that the model compression members have been buckled by the load built up near the end of the foam cutting stroke. The weight of the model body, including the fins which provide stability during the vertical drops, is 33 lb; and the weight of the energy absorbing system, including everything below the thrust ring, is 1-2/3 lb, or very close to 5 per cent of the model body weight. The total model weight, divided by the combined bearing area of the three foam pads, is 0.3 lb per sq in.

The model body, with its fins, cannot be regarded as a scaled model of any prototype vehicle. At a scaling of roughly 10 to 1, however, it can be considered as crudely representative of the Lunar Excursion Module, or LEM, class (see appendix A for scaling).

Drop-Test Results and Discussion

In Fig. 9, a limited set of drop-test results is presented for the model just described. The solid line is the beginning of an envelope extending from the α axis to the V axis, where α is the landing surface slope in degrees and V , the vertical impact velocity in feet per second. The envelope defines the boundary between acceptable and unacceptable impacts referred to earlier. Within the envelope, the model does not tip over, and

the vertical acceleration does not exceed 50 g (which scales to 5 g for a prototype with 10 to 1 scaling and 8 g with 6 to 1 scaling, as shown in Eq (A6) of appendix A). All the data are for a model oriented with one leg uphill and are expressed in terms of symbols as described in Fig. 9.

Now, the end point on the α axis, at $\alpha = 8-1/2^\circ$, is the stability point when the model is released with only the uphill foam touching the ground. The end point on the V axis, which has not been determined, will be the point at which the system goes above 50 g on a horizontal surface. Because of inaccuracies in the accelerometer measuring system, the highest velocity for which the acceleration is known not to exceed 50 g is 32 ft per sec, the maximum velocity for which successful drops are shown in Fig. 9. It is also known, however, that the system did go over the g limit on a horizontal surface at a vertical impact velocity of 44 ft per sec. Thus the end point on the V axis must lie between 32 and 44 ft per sec.

When these velocities are squared and divided by a denominator consisting of 2 g times the ratio of the system weight to the vehicle weight, the results define limits for the over-all efficiency of energy absorption. Hence the lower velocity of 32 ft per sec, together with the weight ratio given earlier as 0.05, gives a conservative estimate of the present efficiency, namely, 320 ft-lb per lb.

A question remains, however, as to whether such an efficiency is reasonably high. In this regard, only the most qualitative statements can be made since no consistent standard has been established for the comparison of efficiencies. As a start, then, on a qualitative comparison, it is noted that at least one of the competitive systems referenced earlier, and having satisfactory tip-over stability, has a lower weight ratio than the present value of 0.05, specifically, 0.025 (ref. 12). In addition, the present value

would have to be increased somewhat if the proof tests were extended to include the effects of horizontal velocity and nonvertical attitude at impact, but were left unchanged in other respects. This constitutes, of course, a weight disadvantage for the present system. Such a disadvantage seems roughly balanced, however, by the fact that 32 ft per sec is larger by a factor of three than the maximum resultant impact velocity usually considered in designing complete systems (refs. 12 and 48). Despite margins of safety in the competitive designs, this velocity factor should remain large enough, particularly when squared, to provide a major advantage in total energy. On balance, then, it seems likely that the present system has a reasonably high over-all efficiency.

The remaining issues to be considered, for the substantiation of the premise in question, involve the satisfaction of the supplementary requirements. With respect to the requirement of an efficient shape for the envelope in the $V-\alpha$ plane, the desired envelope is shown in Fig. 9 as the larger of the two dashed rectangles. Such a rectangle, with side lengths based on the end points just described, is efficient in the sense of being equally restrictive at every impact velocity and ground slope. The actual envelope, as represented by the solid line, is seen to follow the vertical leg of the desired envelope with deviations which are small enough to retain an efficient shape, as required.

It is also noteworthy in Fig. 9 that the deviations of the actual envelope permit a rectangular placard, or permissible landing envelope, with end points at $V = 28$ ft per sec and $\alpha = 5^\circ$. This placard indicates that the supplementary requirement of preventing vehicle tip-over has been met, in a qualitative sense, since a 5° slope is a common lunar module design specification (refs. 12 and 48), albeit currently outdated.

The three remaining supplementary requirements, which are related to packaging, penetration of the lunar crust, and a stable firing platform for return blast-off, have been met in varying measure by the selection of plastic foam and a three-legged system, as pointed out earlier. This completes the generally qualitative substantiation of the premise that an over-all energy absorbing system can be developed which readily incorporates the present supplementary requirements while retaining reasonably efficient energy absorption with respect to stable systems developed previously.

Several of the supplementary requirements referred to in the result just stated have been satisfied because of the foam-cutting mechanism. Unfortunately, this mechanism has a very low SEA value, specifically, 688 ft-lb per lb. A reasonably high over-all efficiency has been achieved despite the low SEA for two reasons. First, the foam mechanism is well suited to the specified over-all design geometry, and it was indicated in the first major section of the paper that in such cases a relatively inefficient mechanism can result in surprisingly good over-all efficiency. The second reason is the use of a deforming support structure, which improves efficiency partly by increasing the energy absorption but mostly by decreasing the over-all weight.

Since these two reasons are expected to be important and relatively unavoidable factors in many designs, the present result should be applicable to a variety of configurations and a variety of supplementary requirements. This means that in the selection of a mechanism for any specific landing problem, a trade-off between a high SEA and those properties which tend to meet important supplementary requirements need not always result in prohibitive losses, or even any losses, in over-all efficiency. Hence a trade-off of this nature may well be feasible with respect to efficient energy absorption as well as

desirable with respect to the supplementary requirements, and such a conclusion manifestly diminishes the significance of the SEA in the selection mechanisms.

MATERIALS NEEDS

The preceding two major sections of the paper have both led to observations tending to downgrade the SEA parameter with respect to its importance in mechanism selection. It follows, then, that any program for materials improvement must include mechanisms having a wide range of SEA values. Because of this range, any comments on a materials program must necessarily be general, and specific materials must be considered only as examples. The present section is devoted to such comments and examples. It starts with a general discussion of the nature of needed materials improvements, and ends with a brief consideration of the possible effect of those improvements.

Nature of Needed Materials Improvements

One of the improvements recommended is a general increase in mechanism SEA values across the broad range referred to above. The earlier downgrading of the importance of the SEA in the selection of mechanisms does not, per se, diminish its importance in the improvement of a given mechanism. When a given mechanism, having a fixed weight ratio of deforming to nondeforming material, is incorporated in a particular system to absorb a specified energy, an increase in the SEA value always lowers the mechanism weight. In turn, this decrease in the mechanism weight always results in a more efficient system, barring the unlikely possibility that the decrease is accompanied by changes in the mechanism size which require a heavier supporting structure. Thus it is generally desirable to increase the SEA

value of any given mechanism until the mechanism weight becomes an essentially vanishing part of the total system weight.

The SEA limit just stated, where the mechanism weight becomes trivial, has interesting implications when it is illustrated numerically. For example, if acceptable over-all efficiencies for manned landing systems range very roughly from 200 to 450 ft-lb per lb, and if all mechanism material is deformed, then an increase in SEA from, say 30,000 ft-lb per lb to infinity, would net at most a 1-1/2 per cent decrease in total system weight. This means, in view of the fact that the total system weight is only 2-1/2 to 5 per cent of the landing weight, that there is relatively little advantage in raising SEA values above 30,000 ft-lb per lb; and this value has already been attained for some mechanisms (refs. 15 and 27). When these numbers are considered together with the fact that the mechanism having the highest SEA is not necessarily the best choice, the combination strongly indicates that the range of mechanisms for which the SEA can most profitably be increased does not include the most efficient mechanisms currently available. It should be emphasized that the conclusion just stated applies primarily to manned landing vehicles.

By definition, the SEA value of a mechanism can be increased by increasing the ratio for which the numerator is the stroke times the average force and the denominator is the weight of the deforming material. By analogy, it seems likely that the SEA value of a mechanism can be increased by changing its material so as to increase the ratio for which the numerator is the maximum permissible strain times the effective stress during deformation and the denominator is the material density.

Examples of this material ratio are provided by the reversing tube and honeycomb core mechanisms considered earlier, and for these mechanisms the

effective stress during deformation is simply the effective yield stress. The formula given earlier for the reversing tube SEA shows the effective yield stress and the material density to enter explicitly according to the material ratio under consideration (see Fig. 1), and this is also the case for the honeycomb core. For both mechanisms, the maximum permissible strain enters implicitly in the sense that an increase in ductility permits the mechanisms to function at larger maximum values of t/D or t/s , and the curves presented earlier indicate that these larger maximums result in an SEA increase.

For more complicated mechanisms, such as foam cutting or a frangible tube, the strain and stress in the present material ratio would probably have to be artificial. No change would be required, however, in the definition of material density, and it should continue to enter the SEA formula explicitly.

In addition to increasing SEA, it would be desirable to decrease elastic bounce-back for a variety of mechanisms. This would improve tip-over stability and thus permit less outreach and a lighter over-all system. In some cases it may be profitable to decrease bounce-back until the mechanism accounts for no more than, say, 10 per cent of the bounce-back stroke of the entire landing vehicle.

By way of illustration, it would seem that a plastic foam could be made to be brittle and thus have less bounce-back than the material used in the present study - if this has not already been done. Specifically, if the model discussed earlier had not required cutting of the foam to reduce bounce-back and prevent tip-over, then the foam could have been crushed at an SEA three times greater than that of the cutting.

Finally, it would be useful to increase the resistance of energy absorbing mechanisms to the environment imposed by space and the rocket exhaust,

namely, heat, cold, vacuum, radiation, micrometeoroid bombardment, and planetary atmospheres. Admittedly, certain partial fixes are available, such as covering the mechanism in an appropriate manner and lengthening the rocket motor until the exhaust misses the mechanism, with the long motor being used for part of the impact energy absorption. It would be an advantage, however, not to have to rely on these partial fixes, and the mechanisms should be improved accordingly.

Possible Effect of Materials Improvements

At this point, it is useful to consider the possible effect of the materials improvements which have been recommended. In view of the broad SEA range over which mechanisms should be improved, the general effect should be a large increase in the number of highly desirable mechanisms. Specifically, there should be a tendency to concentrate SEA improvement on mechanisms having relatively low SEA values since the most efficient mechanisms have reached the point of diminishing returns for manned landings.

CONCLUDING REMARKS

Two specific examples, one analytical and the other experimental, have been used to show that the SEA is not necessarily the primary factor in the selection of energy absorbing mechanisms. Hence, it has been stated that any program for materials improvement must include mechanisms having a wide range of SEA values. Across this range, the recommended improvements have included a major increase in SEA, a major decrease in elastic bounce-back, and an improved resistance to the environment imposed by space and the rocket exhaust. For the case of manned landings, it has been pointed out that there is little advantage in improving the SEA of the most efficient

mechanisms; and thus the recommended materials improvements should tend to raise the lower SEA values more than the higher values.

It then follows as a corollary that such a selective improvement of SEA values should permit increased consideration of landing requirements other than energy absorption, as, for example, packaging, tip-over stability, and prevention of excessive penetration of the lunar crust. The final choice between over-all systems designed to incorporate such supplementary requirements may well depend on which mechanisms benefit most from improved materials.

APPENDIX A

SCALING

The scaling used herein is derived from two conditions: (1) the model is scaled geometrically relative to the prototype; and (2) the model is made from the same material as the prototype, which requires identical stresses and identical material densities. From condition (1), with N taken as a fixed number,

$$\frac{L_p}{L_m} = N, \quad \frac{A_p}{A_m} = N^2, \quad \frac{v_p}{v_m} = N^3 \quad (A1)$$

where L , A , and v are length, area, and volume, respectively, and where the subscripts p and m stand for prototype and model. From the identical stress aspect of condition (2),

$$\frac{F_p}{A_p} = \frac{F_m}{A_m}, \quad \frac{F_p}{F_m} = N^2 \quad (A2)$$

where F is force. From the identical material density aspect of condition (2),

$$\frac{M_p}{v_p} = \frac{M_m}{v_m}, \quad \frac{M_p}{M_m} = N^3 \quad (A3)$$

where M is mass.

An immediate question arises as to whether the weight force W is consistent with Eq (A2). For this force, with Eq (A3),

$$\frac{W_p}{W_m} = \frac{M_p}{M_m} \frac{g_p}{g_m} = N^3 \frac{g_p}{g_m} \quad (A4)$$

where g is acceleration due to gravity. Hence, for consistency with

Eq (A2), it must follow that

$$\frac{g_p}{g_m} = \frac{1}{N} \quad (A5)$$

If the prototype is to land on the moon and the model is to be tested on earth, Eq (A5) requires that $N = 6$ for exact scaling of the weight force.

Other quantities follow directly from Eqs (A1) to (A3) in conjunction with specific laws of nature. With Newtons second law, for example, Eqs (A2) and (A3) yield

$$\frac{a_p}{a_m} = \frac{F_p}{F_m} \frac{M_m}{M_p} = N^2 \frac{1}{N^3} = \frac{1}{N} \quad (A6)$$

where a is acceleration. It should be noted that Eqs (A5) and (A6) both deal with accelerations and both have the same form. This similarity of form must be regarded as a fortunate coincidence since the two equations serve different functions, with Eq (A6) scaling accelerations according to any number N and Eq (A5) specifying N according to the physical accelerations due to gravity.

For scaling the dimensions of velocity V , the formula for constant acceleration $V^2 = 2aS$ can be used, where S is distance over which constant acceleration takes place. With Eqs (A1) and (A6), it is seen that (with general length, L , specialized as S)

$$\frac{V_p^2}{V_m^2} = \frac{a_p S_p}{a_m S_m} = \frac{1}{N} N = 1 \quad (A7)$$

For time scaling another formula for constant acceleration, $V = at$, can be used, where t is time. Eqs (A6) and (A7) then give

$$\frac{t_p}{t_m} = \frac{V_p}{a_p} \frac{a_m}{V_m} = N \quad (A8)$$

Equations (A1) through (A8) have been checked by other physical laws and relations, including work-energy, impulse-momentum, torque-inertia, and stress-bending moment. The results have been satisfactory in that the laws and relations have been found to have the same scaling on both sides of their equations.

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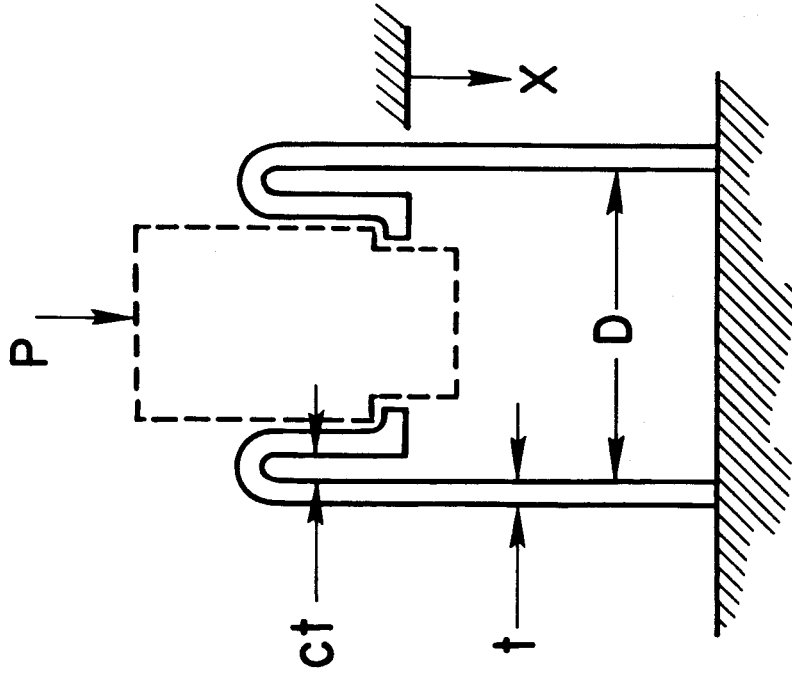
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FIGURE LEGENDS

- Fig. 1.- Reversing of aluminum tube.
- Fig. 2.- Efficiency of reversed tube.
- Fig. 3.- Specific energy absorption.
- Fig. 4.- Hypothetical lunar lander configuration.
- Fig. 5.- Lunar strut configuration.
- Fig. 6.- Weight of struts.
- Fig. 7.- Energy absorbing system.
- Fig. 8.- Test facility and model after impact.
- Fig. 9.- Results on energy absorption and tip-over stability.



ENERGY ABSORBED BY:

1. PLASTIC BENDING
2. PLASTIC COMPRESSION
3. PLASTIC SHEARING
4. FRICTION

$$\text{IF } U = U_{\text{BEND}} + U_{\text{COM}}$$

$$U = \frac{\sigma_y}{(1+c)} \left(\frac{\pi \times D t}{2} \right) + \sigma_y \pi \times (1+c) t^2 = P \times$$

$$\therefore P/\sigma_y = \frac{\pi}{2(1+c)} D t + (1+c) \pi t^2$$

Fig. 1.- Reversing of aluminum tube.

$$\frac{\text{SPECIFIC ENERGY ABSORPTION}}{\text{ABSORPTION}} = \frac{\text{ENERGY ABSORBED}}{\text{WEIGHT MATERIAL DEFORMED}}$$

FOR BENDING AND COMPRESSION:

$$\frac{SEA}{\sigma_y} = \frac{2}{\rho} (1+C) \left(\frac{t}{D} \right) + \frac{1}{\rho(1+C)}$$

$$\text{IF: } \frac{\partial}{\partial C} \left(\frac{SEA}{\sigma_y} \right) = 0; \quad C = \sqrt{\frac{D}{2t}} - 1$$

$$\text{THEN: } \frac{SEA}{\sigma_y} = \frac{2}{\rho} \sqrt{2 \left(\frac{t}{D} \right)}; \quad \sigma_y = K \sqrt{3}$$

Fig. 2.- Efficiency of reversed tube.

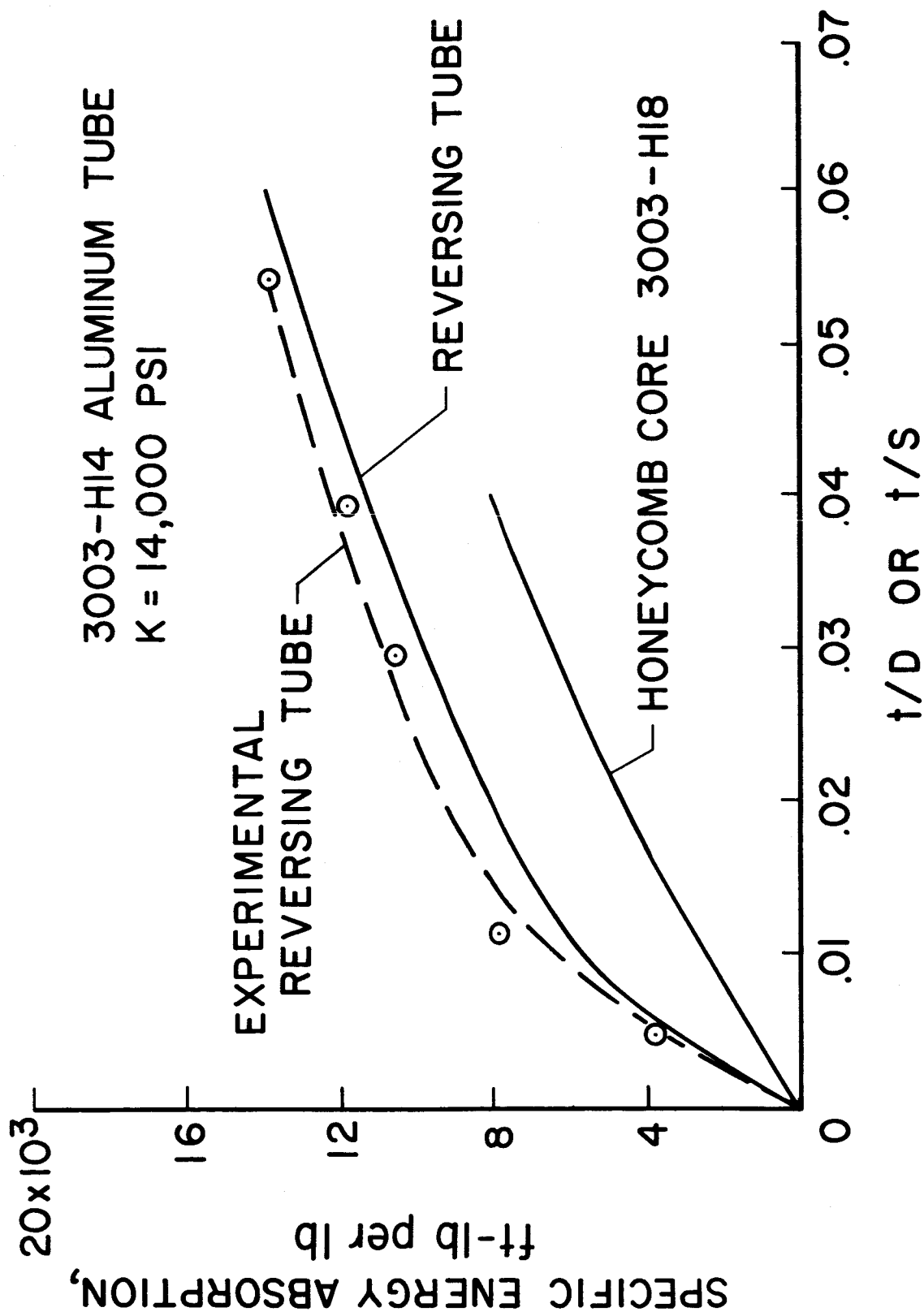
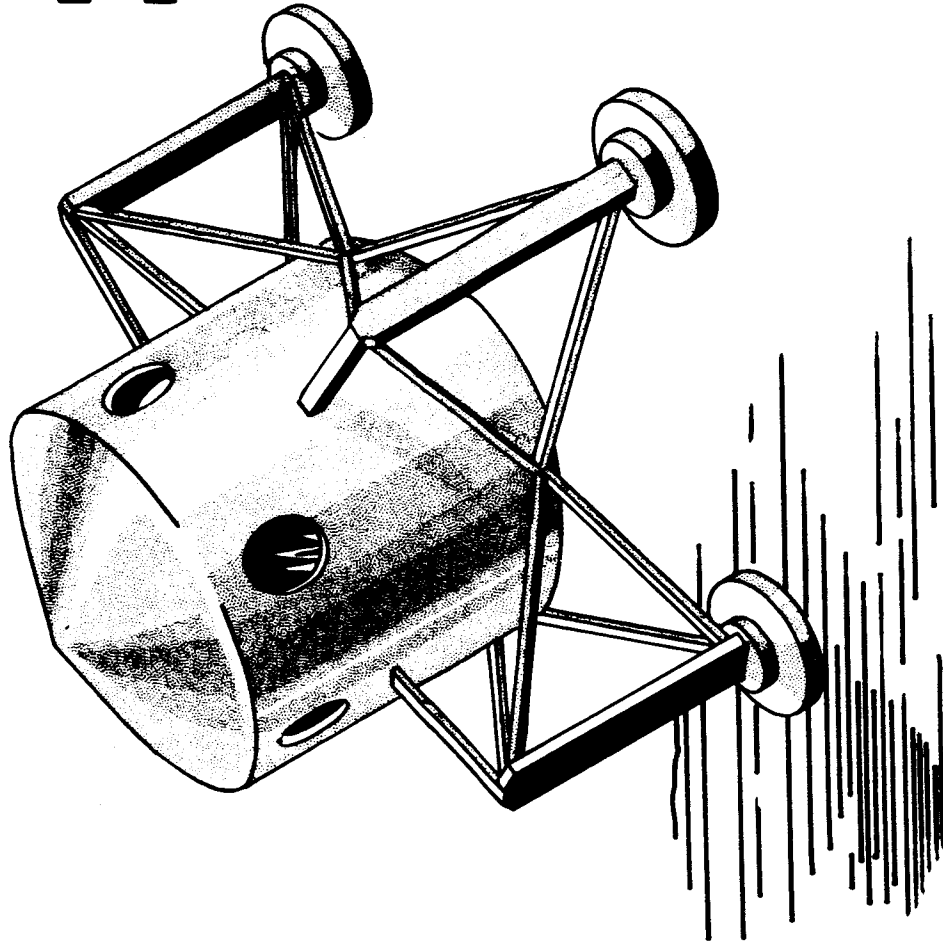


Fig. 3.- Specific energy absorption.



EARTH WEIGHT 20,000 LB

MAXIMUM ACCELERATION:

6 EARTH g

4 MAIN ENERGY

ABSORBING STRUTS

IF ALL 4 PADS IMPACT

SIMULTANEOUSLY:

P = 30,000 LB

Fig. 4.- Hypothetical lunar lander configuration.

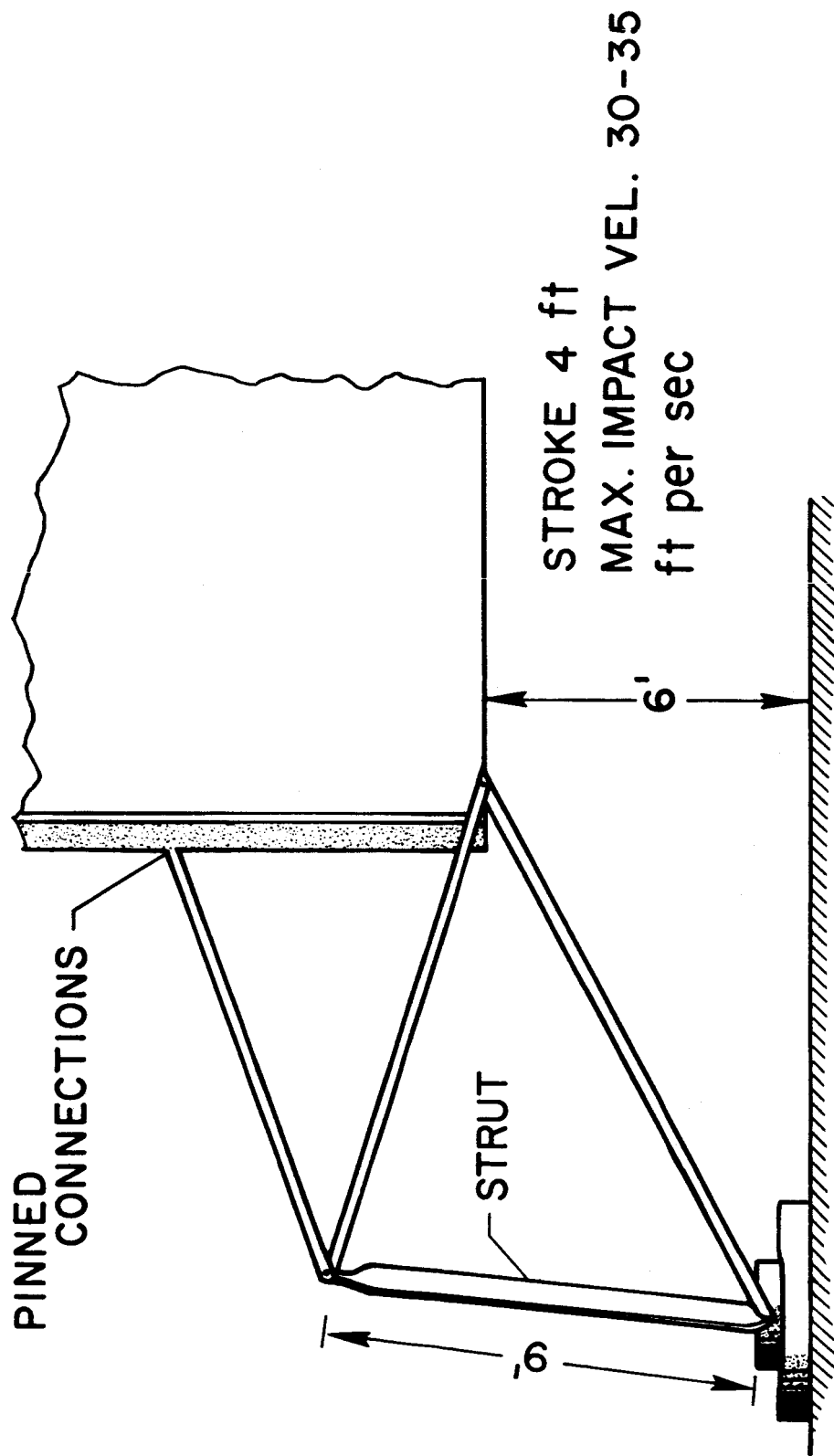


Fig. 5.- Lunar strut configuration.

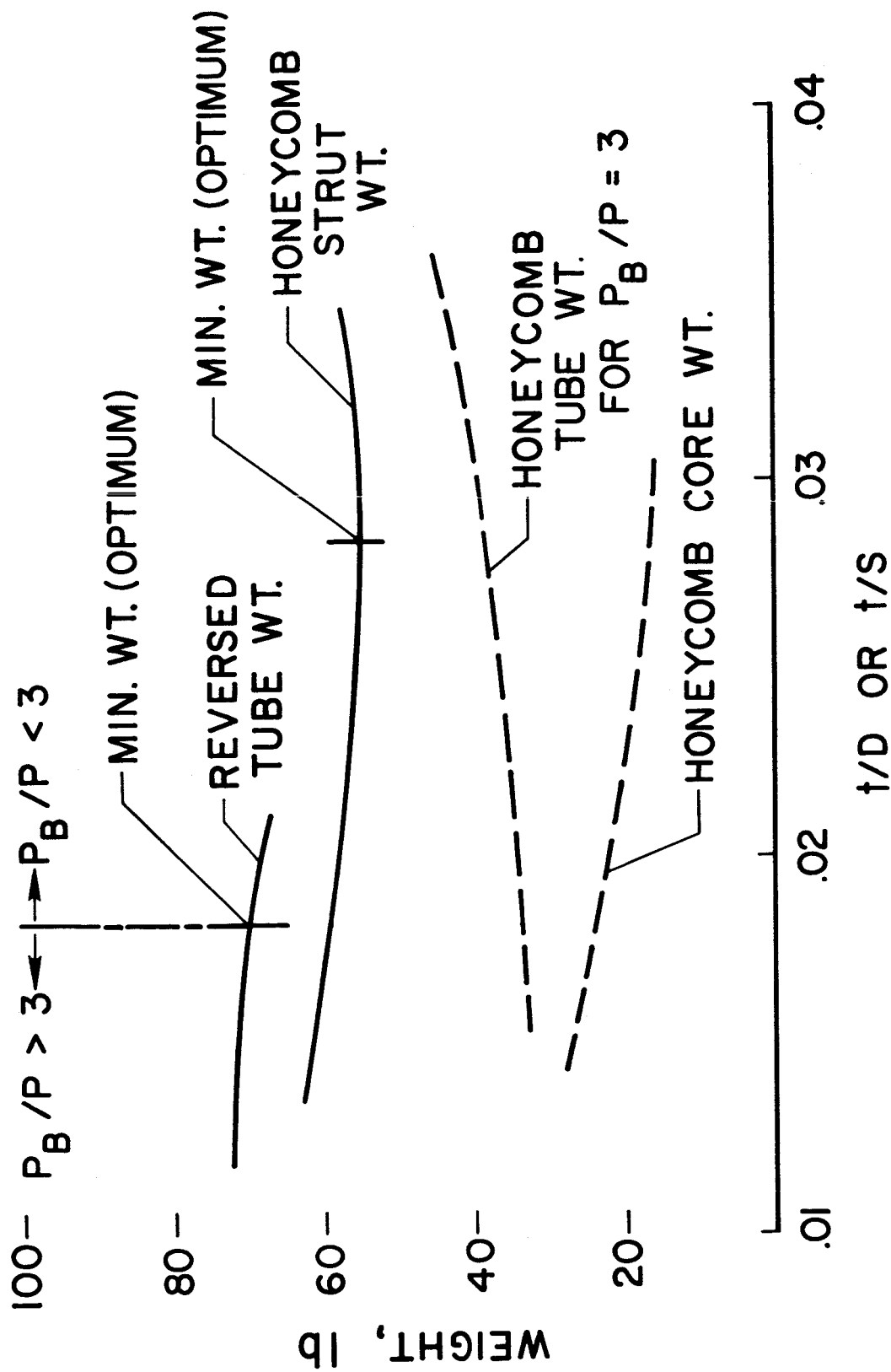
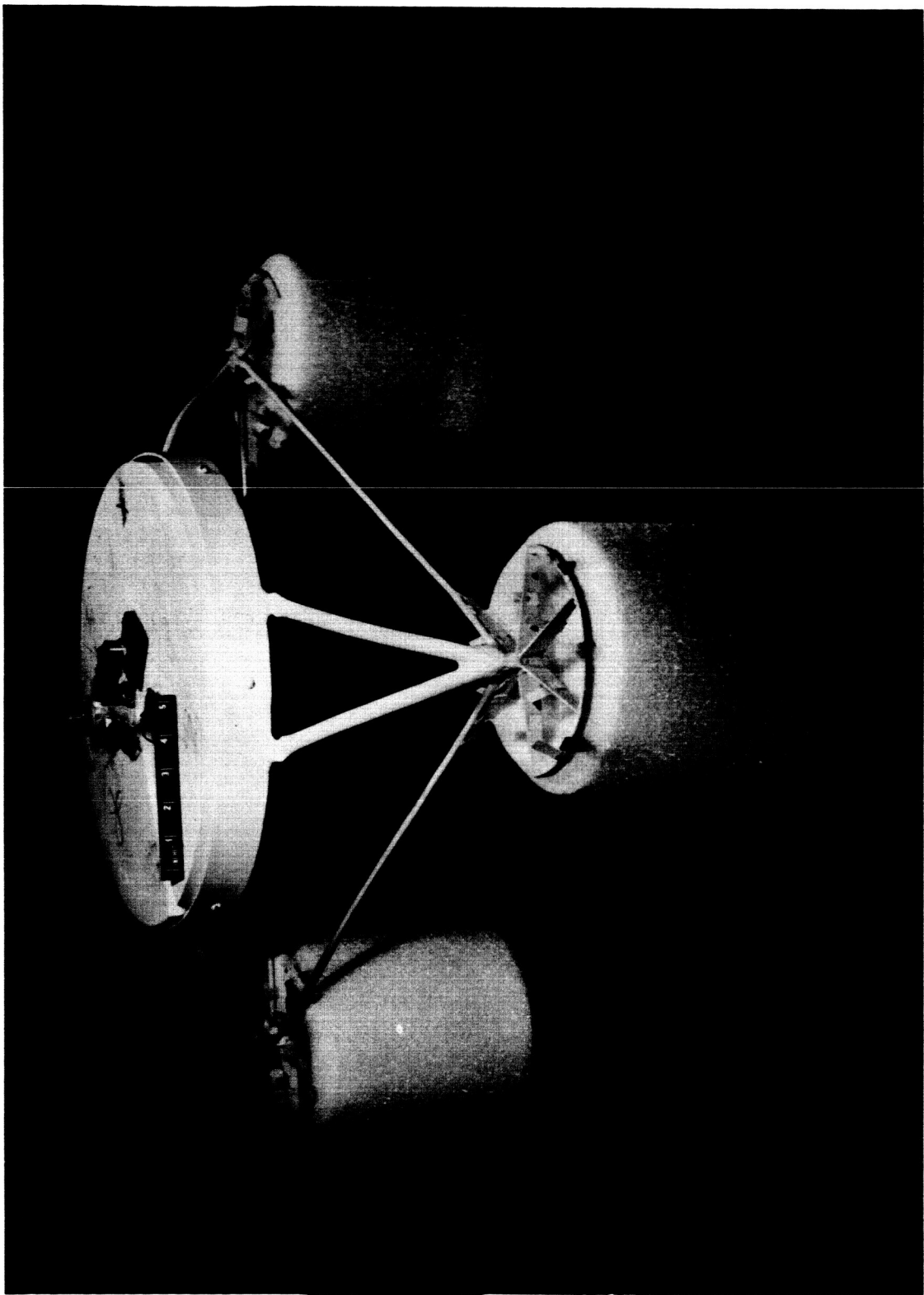


Fig. 6.- Weight of struts.



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Fig. 7.- Energy absorbing system.

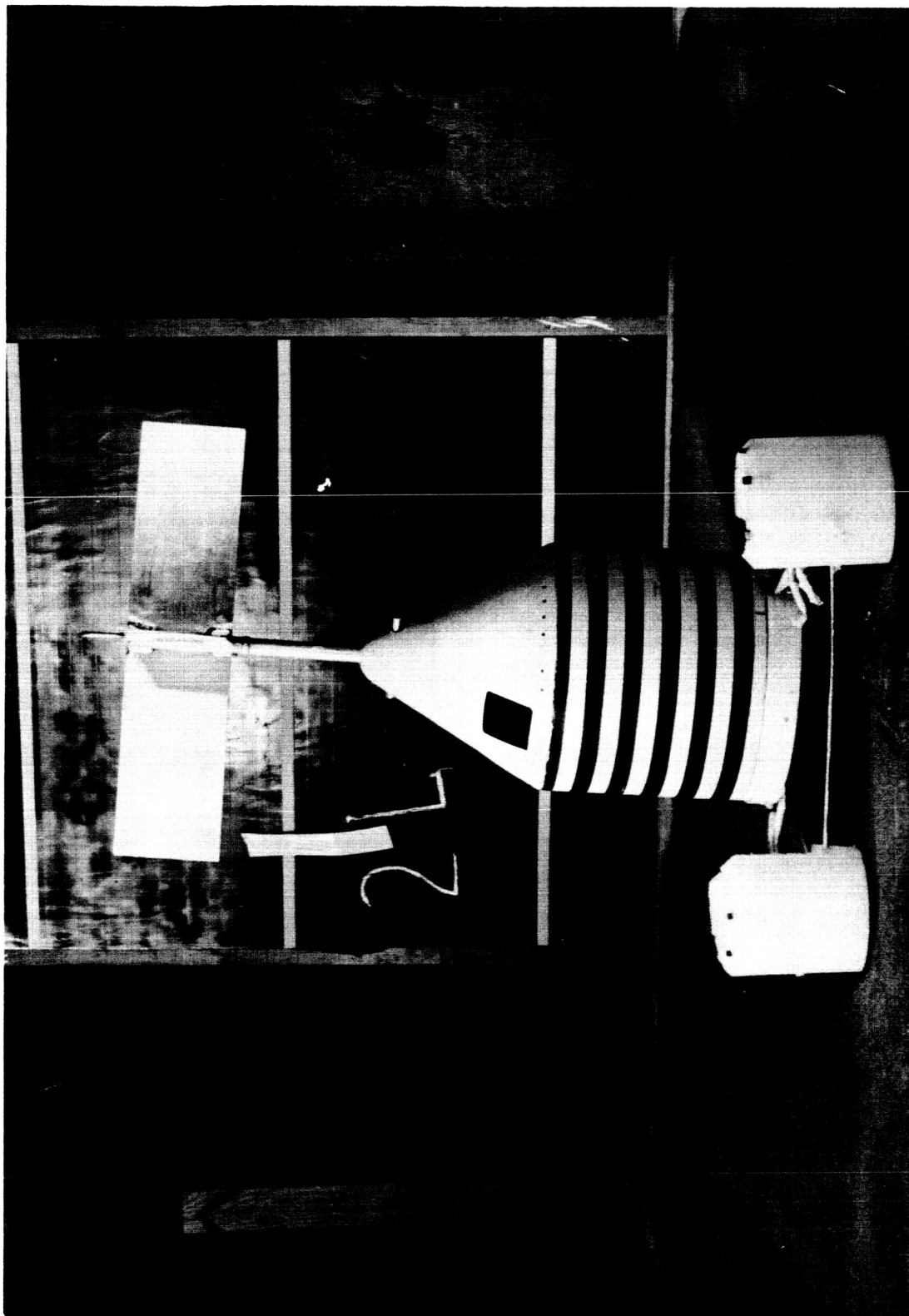


Fig. 8.- Test facility and model after impact.

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